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Calculation, Design, and Experimental Verification of a Resonant Ultrasonic Horn-Workpiece System for Ultrasonic Vibration-Assisted Hard Milling of 90CrSi Steel

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Abstract
Hard milling of hardened steel is commonly associated with high cutting forces, elevated cutting temperatures, rapid tool wear, and difficulty in maintaining low surface roughness. One effective approach to overcoming these limitations is to superimpose ultrasonic vibration on the cutting process; however, a prerequisite for its successful implementation is the design of a vibration transmission system capable of stable operation at resonance. This paper presents the calculation, design, fabrication, and experimental verification of an ultrasonic horn–workpiece system generating axial vibration along the Z direction at a nominal frequency of 20 kHz for application in ultrasonic vibration-assisted hard milling. By selecting 90CrSi tool steel as the material for both the horn and the workpiece, the material properties, including density (ρ) and Young's modulus (E), were determined and used to calculate the acoustic wave velocity and wavelength, thereby establishing the half-wavelength length of the horn–workpiece assembly and the corresponding resonance condition. After fabrication, the system was characterized through impedance scanning and frequency measurement to verify its resonant behavior, while the axial vibration amplitude at the workpiece end face was also measured. The results show that the developed system achieved near-resonant operation at 20 kHz, and that the working amplitude could be adjusted through the design and calibration of the workpiece length Lp to satisfy the amplitude range required for the design of ultrasonic vibration-assisted milling experiments. The study therefore provides a practical workflow of rapid calculation, fabrication, experimental verification, and amplitude tuning that is well suited to laboratory conditions and to the localization of ultrasonic-assisted machining systems.

Keywords: Ultrasonic Horn, 20 kHz Resonance, Amplitude Amplification, 90CrSi Steel, Axial Vibration, Hard Milling, Vibration Amplitude Measurement

1. Introduction

In sectors such as mold and die manufacturing, precision engineering, and other high-technology industries, the demand for machining hard and ultra-high-strength materials has increased rapidly, thereby intensifying the challenges associated with cutting force, vibration, cutting temperature, and surface quality when conventional milling is employed [1-8]. Within this context, vibration-assisted machining has been recognized as a family of hybrid manufacturing technologies capable of improving tool–workpiece contact conditions through the superposition of low-amplitude, high-frequency oscillations, thereby generating an intermittent cutting effect and enhancing the machinability of a wide range of cutting processes [2, 9-12].

With specific regard to ultrasonic vibration-assisted milling, recent review studies have reported clear trends in the improvement of cutting force, surface quality, and process stability, while at the same time emphasizing a common experimental limitation: the ultrasonic system, consisting of the generator, transducer, horn, and load, is highly sensitive to assembly conditions and manufacturing tolerances, and is therefore prone to resonance-frequency deviation in the absence of an appropriate design and verification procedure [13, 14]. Accordingly, horn design should not be regarded merely as the problem of determining the correct half-wavelength length; rather, it constitutes a multi-objective design problem involving amplitude amplification, stress distribution, and vibration-mode stability.

This paper focuses on a fundamental contribution to the implementation of ultrasonic vibration-assisted milling, namely, a systematic procedure for the design, fabrication, and experimental verification of a horn–workpiece system vibrating along the axial Z-direction at 20 kHz on a CNC milling machine, while taking into account assembly constraints and experimental tuning requirements to achieve the target vibration amplitude.

2. Preliminary calculation of the horn–workpiece system length based on the theoretical model

The horn–workpiece ultrasonic vibration system operating at 20.0 kHz must satisfy several essential requirements, including sufficient stiffness and strength, effective ultrasonic transmission, resonance at the operating frequency, low energy dissipation, and stable vibration with an amplitude below 10 µm to ensure suitability for ultrasonic vibration-assisted milling. The horn was designed in the form of a stepped cylindrical horn with a high amplitude amplification ratio [15, 16]. In addition, the structure had to provide adequate space and mechanical integrity for workpiece clamping and mounting on a dedicated fixture.

The overall length of the horn–workpiece system must be equal to an integer multiple of half a wavelength (λ/2) [17-19]. The threaded joint must be of an anti-loosening type suitable for ultrasonic vibration-assisted machining. The total system length can be adjusted through the workpiece length.

According to acoustic wave propagation theory, the sound velocity is calculated as follows:

$$C = \sqrt{\frac{E}{\rho}} \tag{2.1}$$

Where:

- C is the longitudinal wave velocity in the material (m/s),
- E is the Young's modulus (GPa),
- ρ is the density (kg/m³), and
- f is the ultrasonic vibration frequency (Hz).

The density of 90CrSi steel was experimentally determined as follows:

$$\rho_{90CrSi} = 7706\ (kg/m^3) \tag{2.2}$$

The Young's modulus of 90CrSi steel is E = 205 GPa.
The acoustic wave velocity in 90CrSi steel is given by:

$$C = \sqrt{\frac{E}{\rho}} = \sqrt{\frac{205 . 10^9}{7706}} \approx 5157,78\ (m/s) \tag{2.3}$$

The wavelength propagated in 90CrSi steel at a frequency of 20.0 kHz is:

$$\lambda = \frac{C}{f} = \frac{5157,78}{20000} = 0,257888\ (m) = 257,89\ (mm) \tag{2.4}$$

The preliminary length of the horn–workpiece system required to achieve the maximum vibration amplitude is given as follows [20]:

$$L = \frac{C}{2f} = \frac{\lambda}{2} = \frac{257,89}{2} = 128,94\ (mm) \tag{2.5}$$

Where:

- $L = L_1 + L_2$
- $L_1 = L_2 = \frac{L}{2} = \frac{\lambda}{4} = \frac{257,89}{4} = 64,47\ (mm)$

The flange on the horn is used to position and rigidly clamp the horn to the fixture. Therefore, the flange location must be designed at a nodal point of the ultrasonic wave, that is, at the position where the vibration amplitude is zero, in order to avoid energy dissipation and undesired vibrational coupling with the machine structure.

Since the vibration amplitude at the nodal point E is AE = 0, it follows that:

$$Cos\left(\frac{\omega.\ x_E}{C}\right) = 0$$

$$=> \frac{\omega.\ x_E}{C} = 1$$

$$=> x_E = \frac{C}{\omega} = \frac{C}{4f} = \frac{5157,78}{4\ .\ 20000} = 64,47\ (mm)$$

Accordingly, in Fig 1, the horn length of segment L_1 = OE = x_E = 64.47 mm, and point E corresponds to zero vibration amplitude. This location therefore serves as the nodal point for the flange design as Fig 1.

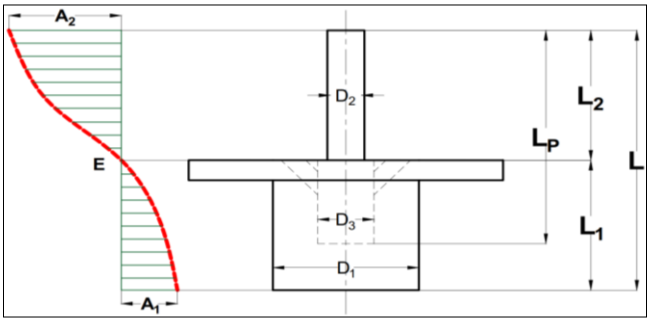

Fig 1: Preliminary calculation model of the horn-workpiece system

In the proposed design, the total length of the horn–workpiece system was defined as $L = L_1 + L_2 = \lambda/2 = 128.94$ mm, with $L_1 = L_2 = 64.47$ mm. The diameters D_1 and D_2 were selected according to practical design requirements, namely $D_1 = 65$ mm and $D_2 = 16.5$ mm. The choice of $D_1 = 65$ mm was made to ensure that the horn was sufficiently large to accommodate a hollow bore of diameter $D_3 = 25$ mm for mounting the workpiece shank, while also providing adequate strength for installation and workpiece clamping during machining, as well as sufficient wall thickness on both sides of the bore to guarantee the reliability and stiffness of the system. The diameter D2 = 16.5 mm is the standard design dimension of a tablet punch head. The target vibration amplitudes at the machining position were achieved by varying the workpiece length, L_p.

The design drawing and the fabricated horn are shown in Fig 2.

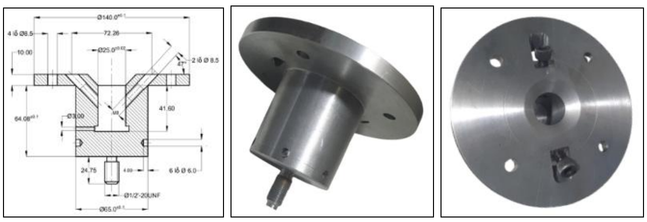

Fig 2: Design drawing of the ultrasonic horn for vibration-assisted milling

3. Calculation and verification of the resonant frequency of the horn–workpiece system

To verify the resonant frequency of the horn–workpiece system, a PicoScope 2000 Series 5-in-1 Portable Oscilloscope (2205A, 25 MHz, 200 MS/s, UK) was employed in combination with a dedicated electronic circuit for waveform observation and indirect impedance estimation of the system, as shown in Fig 3.

Fig 3: Frequency sweeping, impedance measurement, and resonance verification of the horn–workpiece system

The resonance test results of the horn–workpiece system showed that the system exhibited resonant peaks at 20.001 kHz and 20.906 kHz as Fig 4.

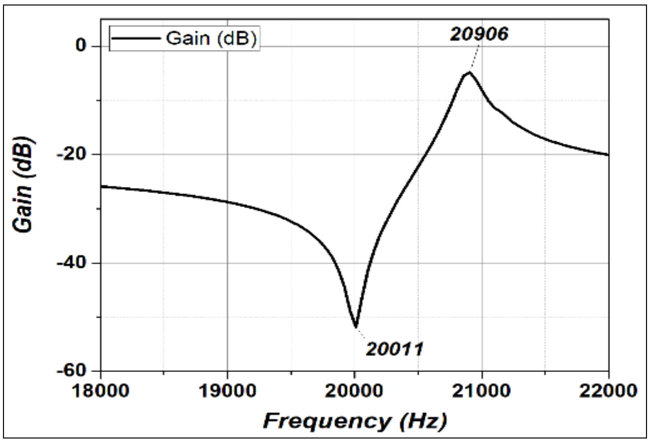

Fig 4: Screenshot of the resonant frequencies of the horn–workpiece system

4. System Configuration

The ultrasonic vibration system consisted of a high-power ultrasonic generator (1) and a commercial transducer used as the excitation source (2). The ultrasonic generator was a commercial MPI unit manufactured in Switzerland, with a rated power of 3000 W and an operating frequency range of 15–100 kHz. The transducer was an RPS-5020-4Z model, manufactured in China, with a rated power of 1500 W, an operating frequency of 20 kHz, and a nominal vibration amplitude of 8 µm. The specifications of both the ultrasonic generator and the transducer were re-measured and experimentally verified, and were subsequently used as the input boundary conditions for the design of the horn–workpiece ultrasonic vibration assembly.

The horn–workpiece assembly, comprising the horn (3) and the workpiece (4), was designed in a stepped cylindrical configuration intended to generate one-dimensional longitudinal vibration. The design principles included accurate resonance at 20 kHz, an overall length equal to an integer multiple of half a wavelength, clamping at the nodal position, the use of a British-standard threaded joint resistant to self-loosening, and control of the contact surface quality to minimize energy loss. The structural configuration of the vibration system is shown in Fig 5.

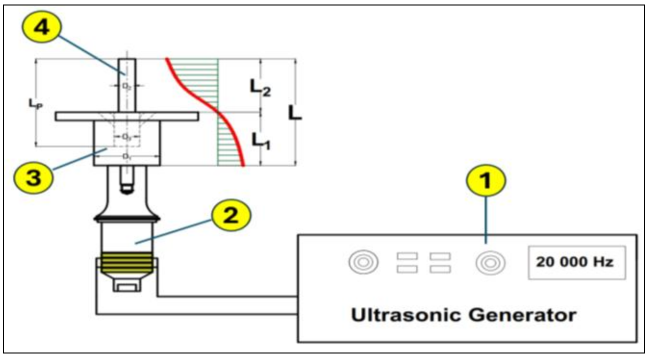

Fig 5: Calculation model of material properties, wavelength, and resonant length

5. Experimental investigation and optimization for determining workpiece length and target amplitude

5.1 Optimization of the horn–workpiece system length

In the present study, the workpiece was a stepped cylindrical tablet punch, the detailed dimensions and fabricated form of which are shown in Fig 6.

Fig 6: Workpiece design drawing and fabricated workpiece

According to the preliminary calculation, the workpiece lengths L_p corresponding to the three target levels were initially estimated as 58.01 mm, 61.4 mm, and 64.7 mm. In order to optimize and experimentally verify the horn–workpiece system length for achieving the three target vibration amplitudes of 4.0 µm, 6.0 µm, and 8.0 µm, an exploratory experiment was carried out to identify both the target-amplitude range and the maximum-amplitude range as a function of workpiece length. A total of 42 workpieces with lengths ranging from L_p = 54 to 136 mm, with an increment of 2 mm between adjacent specimens, were prepared by keeping the head portion of the workpiece constant while varying the shank length.

5.2 Two-step optimization and calibration of workpiece length for achieving the target amplitude

Step 1. *Experimental determination of the workpiece-length ranges corresponding to the target amplitude range and the maximum amplitude range*

The objective of this step was to determine the horn–workpiece system length by varying the workpiece length L_p in order to identify both the maximum vibration amplitude and the target vibration amplitudes at a fixed frequency of 20.0 kHz. Based on the theoretical resonance condition, the target total length of the horn–workpiece system was calculated as L = 128.94 mm.

The axial vibration amplitude along the Z direction of each workpiece was measured using a Philtec D170 fiber-optic displacement sensor (USA), a NI USB-6421 data acquisition device with a sampling rate of 250 kS/s (Hungary), and NI Signal Express 2015 software, with an analysis resolution of up to 1 µm. The experimental arrangement is shown in Fig 7. Each workpiece was sequentially mounted on the ultrasonic vibration system and excited at a constant frequency of 20.0 kHz using an MPI WG ultrasonic generator (Switzerland) and an RPS-5020-4Z transducer (China), while the vibration amplitude at the end face of the workpiece was recorded.

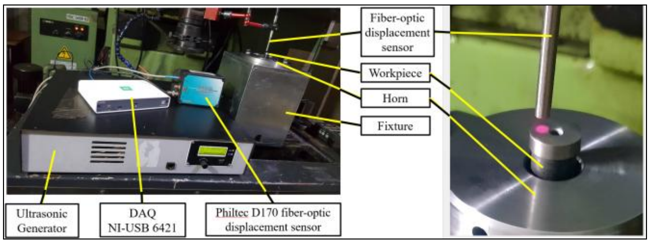

Fig 7: Schematic of the ultrasonic vibration amplitude measurement system

The vibration amplitude of each workpiece was measured three times, and the average value was calculated. The summarized results are presented in Table 1. In addition, the relationship between the total horn–workpiece length and the vibration amplitude is reported in Fig 8. These experimental results of ultrasonic vibration amplitude were subsequently used to determine the optimal workpiece length L_p, so as to ensure not only the attainment of the maximum vibration amplitude, but also the realization of the three target amplitude levels under resonant conditions at the design frequency.

Table 1: Experimental results of vibration amplitude measurement for the horn–workpiece system

STT	L_P (mm)	A (µm)	STT	L_P (mm)	A (µm)	STT	L_P (mm)	A (µm)
1	54,0	1,0	15	82,0	15,5	29	110,0	23,5
2	56,0	2,5	16	84,0	16,0	30	112,0	24,0
3	58,0	3,9	17	86,0	17,1	31	114,0	24,5
4	60,0	4,7	18	88,0	18,0	32	116,0	24,8
5	62,0	6,0	19	90,0	18,5	33	118,0	24,6
6	64,0	6,8	20	92,0	19,0	34	120,0	24,5
7	66,0	7,9	21	94,0	19,5	35	122,0	24,0
8	68,0	8,5	22	96,0	20,0	36	124,0	23,6
9	70,0	9,0	23	98,0	20,5	37	126,0	23,0
10	72,0	10,5	24	100,0	20,7	38	128,0	22,5
11	74,0	11,5	25	102,0	21,5	39	130,0	22,0
12	76,0	13,0	26	104,0	22,0	40	132,0	21,6
13	78,0	13,8	27	106,0	22,5	41	134,0	21,0
14	80,0	14,6	28	108,0	23,0	42	136,0	20,0

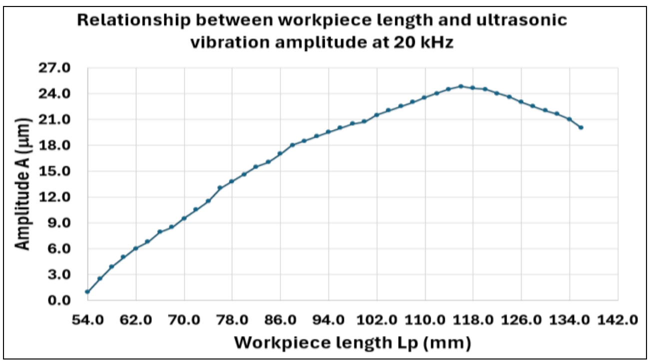

Fig 8: Relationship between ultrasonic vibration amplitude and the total length of the horn-workpiece system

Through the experimental measurement of ultrasonic vibration amplitude, two amplitude ranges corresponding to two workpiece-length intervals were identified as follows:

- Within the workpiece-length range of 110 - 120 mm, the system exhibited a maximum-amplitude region, with the ultrasonic vibration amplitude reaching approximately 24.8 µm.
- Within the workpiece-length range of 56 - 70 mm, the system provided the target-amplitude region corresponding to the three desired amplitude levels of 4.0 µm, 6.0 µm, and 8.0 µm.

Step 2. *Experimental calibration and precise determination of workpiece length for achieving the three target amplitude levels required for the design of experiments*

A second-stage experimental procedure was then conducted to calibrate and accurately determine the workpiece lengths required to achieve both the target amplitudes and the maximum amplitude.

Accordingly, 15 workpieces with lengths ranging from L_p = 56 to 70 mm and 11 workpieces with lengths ranging from 110 to 120 mm were fabricated in the same batch, with an increment of 1 mm between adjacent specimens, for ultrasonic vibration-amplitude measurements. The experimental arrangement, measurement procedure, and instrumentation were identical to those used as Fig 7. The results are presented in Table 2 and illustrated in Figures 9.

Table 2: Experimental results for the target amplitudes and the maximum amplitude

STT	L_p (mm)	A (µm)	STT	L_p (mm)	A (µm)	STT	L_p (mm)	A (µm)
1	56	2,3	10	65	7,5	19	113	23,0
2	57	3,0	11	66	7,9	20	114	23,4
3	58	3,9	12	67	8,5	21	115	23,7
4	59	4,5	13	68	9,0	22	116	24,0
5	60	5,0	14	69	9,5	23	117	23,8
6	61	5,5	15	70	10,3	24	118	23,5
7	62	6,1	16	110	21,6	25	119	23,0
8	63	6,5	17	111	22,1	26	120	22,5
9	64	7,0	18	112	22,6			

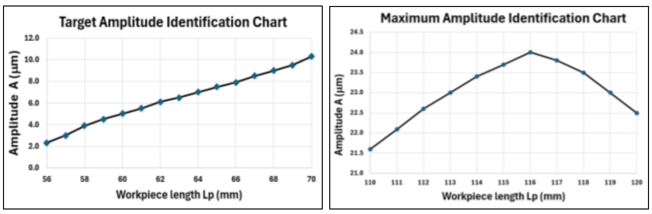

Fig 9: Experimental plot of the maximum amplitude and target amplitudes

In conclusion, the three target vibration amplitudes were determined to be 4 µm, 6 µm, and 8 µm, corresponding to workpiece lengths of L_p = 58.0 mm, 62.0 mm, and 66.0 mm, respectively.

5.3 Discussion

Designing the horn based on the half-wavelength criterion is necessary but not sufficient. Experimental calibration through adjustment of the workpiece length plays a decisive role in achieving the desired working amplitude in vibration-assisted machining, where the amplitude is typically on the order of only a few micrometers. This observation is consistent with the prevailing trend in modern horn-design studies, in which one-dimensional theoretical calculations are first employed to define the primary geometric framework, followed by FEM/FEA analysis and/or experimental verification to refine the frequency–amplitude response under actual assembly and operating conditions [5].

A notable feature of the present study is the one-dimensional vibration transmission configuration applied along the axial Z-direction directly to the workpiece, whereas such configurations appear to be less frequently reported in the literature on vibration-assisted milling than other ultrasonic vibration arrangements. In this respect, the contribution of the present work lies in proposing a systematic procedure that integrates theoretical calculation, structural design, optimization, and experimental validation through resonance verification and calibration under actual clamping and assembly conditions for a one-dimensional axially vibrating horn–workpiece system.

6. Conclusions

This paper proposed and experimentally verified a systematic procedure for the calculation, design, fabrication, and measurement of a 20 kHz resonant horn–workpiece system for ultrasonic vibration-assisted hard milling, in which one-dimensional ultrasonic vibration was applied directly to the workpiece on a CNC milling machine. The material properties of 90CrSi steel provided the basis for determining the wavelength and half-wavelength length required for system design. Resonance measurements confirmed that the developed system operated in the vicinity of 20 kHz. The experimental results further showed that the vibration amplitude at the workpiece end face depended strongly on the workpiece length and could be calibrated to achieve the target amplitudes required for machining experiments. These findings provide a practical foundation for the further development and localization of ultrasonic-assisted machining systems under laboratory conditions.

7. Acknowledgments

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